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(54) **HYDRAULICALLY DRIVING SYSTEM.**

(57) A hydraulically driving system which comprises: a plurality of flowrate detectors (10A, 10B) for respectively detecting flowrates supplied to a plurality of hydraulic actuators (3A, 3B); valve control devices (11A, 11B) for controlling a plurality of flowrate regulating valves (40A, 40B) such that the flowrates detected by the plurality of flowrate detectors coincide with flowrates instructed by a plurality of control levers (5A, 5B); and pump included rotation control devices (12; 12A-12F) for controlling a discharge flowrate from a hydraulic pump such that the discharge flowrate from the hydraulic pump (1) is less than the total sum of the flowrates instructed by the plurality of control levers by a predetermined flowrate (ΔQref; Xref). The pump inclined rotation control device controls the discharge flowrate of the hydraulic pump (1) by use of flowrate deviations (ΔQ1, ΔQ2) obtained by respectively subtracting the flowrates detected by the flowrate detectors (10A, 10B) from the flowrates instructed by the control levers (5A, 5B).

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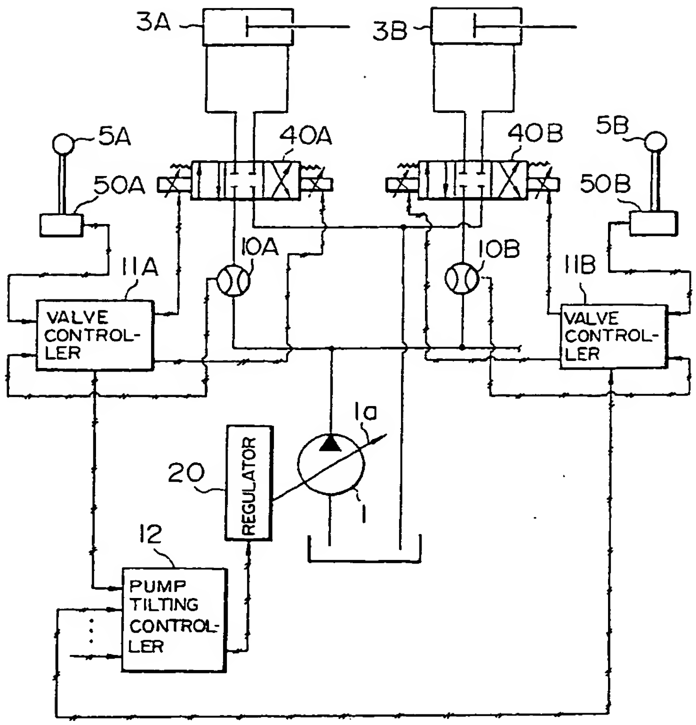
FIG. 1


3A
3B
5A
40A
40B
5B
50A
50B
10A
10B
11A
11B
VALVE CONTROL-LER
VALVE CONTROL-LER
1a
20
REGULATOR
1
12
PUMP TILTING CONTROL-LER

TECHNICAL FIELD

The present invention relates to a hydraulic drive system for driving a plurality of hydraulic actuators by a single variable displacement hydraulic pump, and more particularly to a hydraulic drive system for driving a plurality of hydraulic actuators while controlling a delivery rate of a hydraulic pump depending on a demanded flow rate.

BACKGROUND ART

As to a hydraulic drive system for driving a plurality of hydraulic actuators by a single variable displacement hydraulic pump, there is known a so-called load sensing control system in which a delivery rate of the hydraulic pump is controlled in such a manner as to supply a flow rate only demanded by the hydraulic actuators. The load sensing control system is described in, for example, West German Patent No. 3,321,483, JP, B, 60-11706 and JP, A, 2-261902.

The load sensing control system (hereinafter referred to as an LS control system) comprises a variable displacement hydraulic pump, a plurality of hydraulic actuators connected to the hydraulic pump in parallel, a plurality of flow control valves for respectively driving the plurality of hydraulic actuators, a plurality of control levers for instructing respective flow rates to the plurality of flow control valves, a circuit for detecting maximum one of load pressures of the plurality of hydraulic actuators, and a pump regulator for controlling a delivery rate of the hydraulic pump so that a delivery pressure of the hydraulic pump is held higher a fixed value than the maximum load pressure.

When any one of the control levers is operated, the associated flow control valve is opened with an opening corresponding to an input amount from that control lever (i.e., a demanded flow rate), whereby a hydraulic fluid from the hydraulic pump is supplied to the associated hydraulic actuator through a pressure compensating valve and the flow control valve. Simultaneously, a load pressure of that hydraulic actuator is introduced as the maximum load pressure to the pump regulator which controls the pump delivery rate so that the pump delivery pressure is held higher a fixed value than the maximum load pressure. At this time, when the input amount from the control lever (i.e., the demanded flow rate) is small, the opening of the flow control valve is also small and so is a flow rate of the hydraulic fluid passing through the flow control valve, so that the pump delivery pressure is held higher a fixed value than the maximum load pressure at the small pump delivery rate. When the input amount from the control lever (i.e., the demanded flow rate) is enlarged, the opening of the flow control valve is also increased and so does the flow rate of the hydraulic fluid passing through the flow control valve, whereupon the pump delivery rate is increased to keep the pump delivery pressure higher a fixed value than the maximum load pressure.

Meanwhile, in the system making control of the pump delivery rate in that way, when plural hydraulic actuators are simultaneously driven by operating plural control levers, the flow control valve associated with the hydraulic actuator on the lower load side produces a larger differential pressure across the same than that on the higher load side, and the hydraulic fluid is supplied at a larger flow rate to the hydraulic actuator on the lower load side. The combined operation of those plural hydraulic actuators can no longer be performed in accordance with an opening ratio between the flow control valves (i.e., a demanded flow rate ratio). To prevent such a disadvantage, the LS control system includes a pressure compensating valve disposed upstream of the flow control valve for controlling a differential pressure across the flow control valve. When the differential pressure across the flow control valve associated with the hydraulic actuator on the lower load side becomes large during the combined operation, the upstream pressure compensating valve is operated in a valve-closing direction to restrict the flow rate, thereby reducing the differential pressure across that flow control valve. As a result, the differential pressures across the flow control valves on both the higher and lower load sides are maintained at substantially the same value, enabling the associated plural actuators to be simultaneously driven in accordance with the opening ratio between the flow control valves (i.e., the demanded flow rate ratio).

With the LS control system, as mentioned above, since the delivery rate of the hydraulic pump is controlled depending on the demanded flow rate, a part of the pump delivery rate which is wastefully consumed can be reduced to make economical operation possible. In order to surely perform the combined operation, the pressure compensating valve requires to be provided for controlling the differential pressure across the associated flow control valve.

Relating to the LS control system, particularly, there is also known U.S. Patent No. 4,712,376 which discloses a system that the total of input amounts from all the control levers (i.e., demanded flow rates) is calculated for the purpose of controlling respective openings of the flow control valves. This disclosed system is intended to cope with a lack of the pump delivery rate during combined operation of driving plural actuators, by restricting the respective openings of the flow control valves depending on the amount of such a lack, so that

the combined operation is performed in accordance with a demanded flow rate ratio. In addition, though not directly related to the LS control, JP, A, 52-76585 discloses a system in which a flow rate of the hydraulic fluid supplied to a hydraulic actuator is detected for controlling an opening of an associated flow control valve so that the flow rate is held in match with a demanded flow rate.

DISCLOSURE OF THE INVENTION

However, the above-mentioned LS control system has had the following problems.

In a hydraulic drive system of the type adopting LS control, as explained above, there produces a differential pressure across the flow control valve. Given the differential pressure across the flow control valve being ΔP_1, the differential pressure ΔP_1 is determined by a rated flow rate and size of the flow control valve. If the flow control valve used has a large size relative to its rated flow rate, the differential pressure ΔP_1 can be set to a small value. On the contrary, if the flow control valve used has a small size relative to its rated flow rate, the differential pressure ΔP_1 must be set to a large value. Also, the differential pressure ΔP_1 must be set to a value which is produced when the hydraulic fluid flows at the rated flow rate with the input amount from the control lever maximized to make the opening of the flow control valve maximum. Therefore, in the case of using a flow control valve of which size is small relative to its rated flow rate for reducing the system size, the differential pressure ΔP_1 necessarily becomes a large value.

Additionally, the differential pressure ΔP_1 is not determined by the above conditions only. More specifically, viscosity of working oil (hydraulic fluid) is changed to a large extent depending on temperatures and becomes large at a low temperature. To enable the hydraulic fluid to flow at a rated flow rate even under a low temperature, therefore, it is required that the differential pressure ΔP_1 be set to a higher value with a margin. Accordingly, the value of the differential pressure ΔP_1 must be larger than the value determined by the foregoing conditions. In particular, when the hydraulic drive system is used with a hydraulic machine such as a hydraulic excavator, there is a substantial possibility that the construction machine is used in outdoor environment at an extremely low temperature, which requires the margin to be relatively large and hence renders the differential pressure ΔP_1 more increased.

Thus, the differential pressure ΔP_1 across the flow control valve is usually set to a large value and a pressure loss in the hydraulic circuit also becomes large correspondingly.

Furthermore, the LS control system generally includes the pressure compensating valve as mentioned above. The pressure compensating valve also produces a pressure loss ΔP_2 besides the differential pressure ΔP_1 across the flow control valve. The pressure lossΔP_2 comprises a pressure loss produced by the pressure compensating valve itself (i.e., a pressure loss produced when the pressure compensating valve is maximally opened), and a pressure loss produced due to that the pressure compensating valve associated with the actuator on the lower load side is restricted.

In the LS control system, therefore, the pump delivery rate must be controlled in consideration of the differential pressure ΔP_1 and the pressure loss ΔP_2 so that the pump delivery pressure is held higher a fixed value than the maximum load pressure. State otherwise, assuming that the fixed value in the LS control is a target differential pressure ΔP_0, this target differential pressure ΔP_0 must be set to a value larger than the sum of the differential pressure ΔP_1 and the pressure loss ΔP_2 and, in practice, it is set to a still larger value in consideration of a pressure through lines and so on. The target differential pressure ΔP_0 is usually in a range of 15 to 30 bar and this value cannot be said to be small relative to a usual rated value of the hydraulic circuit in a range of 250 to 350 bar.

Another problem experienced in the LS control system is as follows. As explained above, the flow rate of the hydraulic fluid supplied to the hydraulic actuator is adjusted on condition that the differential pressure across the flow control valve is held constant by the pressure compensating valve. In practice, however, a flow of the hydraulic fluid (working oil) passing through the flow control valve is always affected by viscosity of the working oil. Particularly, when the working oil has high viscosity at a low temperature, the flow rate of the hydraulic fluid supplied to the hydraulic actuator becomes smaller than that instructed by the input amount from the control lever (i.e., the demanded flow rate).

An object of the present invention is to provide a hydraulic drive system which has a function of controlling a delivery rate of a hydraulic pump in accordance with a demanded flow rate, produces a small pressure loss, and can perform high-accurate flow control regardless of temperatures of working oil.

To achieve the above object, according to the present invention, there is provided a hydraulic drive system comprising a variable displacement hydraulic pump, a plurality of hydraulic actuators connected to said hydraulic pump in parallel, a plurality of flow control valves for respectively driving said plurality of hydraulic actuators, and a plurality of flow rate instructing means for instructing

respective flow rates to said plurality of flow control valves, wherein said system further comprises a plurality of flow rate sensor means for detecting respective flow rates supplied to said plurality of hydraulic actuators, first control means for respectively controlling said plurality of flow control valves so that the flow rates detected by said plurality of flow rate sensor means are coincident with the flow rates instructed by said plurality of flow rate instructing means, and second control means for controlling a delivery rate of said hydraulic pump such that the delivery rate of said hydraulic pump is smaller by a predetermined flow rate than the total of the flow rates instructed by said plurality of flow rate instructing means.

In the above hydraulic drive system, preferably, said second control means controls a displacement volume of said hydraulic pump such that the total of the flow rates detected by said plurality of flow rate sensor means is smaller by said predetermined flow rate than the total of the flow rates instructed by said plurality of flow rate instructing means.

Also, in the above hydraulic drive system, preferably, said second control means controls the delivery rate of said hydraulic pump by using flow rate deviations resulted from respectively subtracting the flow rates detected by said plurality of flow rate sensor means from the flow rates instructed by said plurality of flow rate instructing means.

Further, in the above hydraulic drive system, preferably, said second control means comprises first calculation means for calculating the total of flow rate deviations resulted from respectively subtracting the flow rates detected by said plurality of flow rate sensor means from the flow rates instructed by said plurality of flow rate instructing means, deviation output means for outputting a value corresponding to said predetermined flow rate as a reference deviation, second calculation means for calculating a difference between the total of the flow rate deviations obtained by said first calculation means and the reference deviation output from said deviation output means, and third calculation means for determining a target displacement volume of said hydraulic pump based on the difference obtained by said second calculation means. In this case, said first calculation means preferably comprises means for adding said flow rate deviations. Said first calculation means may comprise means for selecting a maximum value of said flow rate deviations.

Moreover, in the above hydraulic drive system, preferably, said second control means comprises first calculation means for calculating the total of the flow rates instructed by said plurality of flow rate instructing means, deviation output means for outputting a value corresponding to said predetermined flow rate as a reference deviation, second calculation means for calculating a difference between the total of the instructed flow rates obtained by said first calculation means and the reference deviation output from said deviation output means, and third calculation means for determining a target displacement volume of said hydraulic pump based on the difference obtained by said second calculation means.

Additionally, in the above hydraulic drive system, preferably, said second control means includes deviation output means for outputting a value corresponding to said predetermined flow rate as a reference deviation. Said deviation output means preferably stores said reference deviation as a constant beforehand. Said deviation output means may include means for determining said reference deviation depending on the total of the flow rates instructed by said plurality of flow rate instructing means. Also, said deviation output means may include means for determining one of said plurality of hydraulic actuators which is subjected to a maximum load pressure, means for selecting one of the flow rates instructed by said flow rate instructing means which corresponds to said hydraulic actuator subjected to the maximum load pressure, and means for determining said reference deviation depending on said selected instructed flow rate.

Furthermore, in the above hydraulic drive system, preferably, said second control means comprises integration means for calculating a target displacement volume of said hydraulic pump adapted to make the delivery rate of said hydraulic pump smaller by said predetermined flow rate than the total of the flow rates instructed by said plurality of flow rate instructing means, means for calculating the total of the flow rates instructed by said plurality of flow rate instructing means, means for calculating a modification value for said target displacement volume based on the total of said instructed flow rates, and means for adding said modification value to the target displacement volume calculated by said integration means and calculating a final target displacement volume.

In the present invention thus arranged, the first control means performs flow servo control such that the flow rates detected by the flow rate sensor means are coincident with the flow rates instructed by the flow rate instructing means. Through this flow servo control, the hydraulic actuators are always supplied with the hydraulic fluid (working oil) at respective flow rates corresponding to the instruction values from the flow rate instructing means in spite of change in temperatures of the working oil, etc. The second control means controls the delivery rate of the variable displacement hydraulic pump such that the delivery rate of the

hydraulic pump is smaller by the predetermined flow rate than the total of the flow rates instructed by the flow rate instructing means. By so controlling the pump delivery rate to become smaller by the predetermined flow rate, it is possible with the above flow servo control that the flow control valve associated with the hydraulic actuator producing the maximum load pressure is controlled to be maximized in its opening, and hence a pressure loss produced by that flow control valve can be reduced.

By effecting the above control of the pump delivery rate by the second control means using flow rate deviations resulted from respectively subtracting the flow rates detected by the flow rate sensor means from the flow rates instructed by the flow rate instructing means, an influence of errors in the flow rate sensor means, control equipment for the hydraulic pump and so on can be eliminated and the aforesaid predetermined flow rate can be set to a small value when the pump delivery rate is to be controlled in accordance with demanded flow rates in parallel to the above flow serve control. As a result, an amount of deficiency in the flow rate supplied to the hydraulic actuator producing the maximum load pressure can be made smaller to enable accurate flow control.

By effecting the above control of the pump delivery rate by the second control means using the calculated total of the flow rates instructed by the flow rate instructing means, pump delivery rate can be controlled independently of the flow servo control, which enables stable control free from hunting.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is a diagram of a hydraulic drive system according to a first embodiment of the present invention.

Fig. 2 is a block diagram showing a function of a valve controller shown in Fig. 1.

Fig. 3 is a block diagram showing a function of a modification of the valve controller shown in Fig. 1.

Fig. 4 is a block diagram showing a function of a pump tilting controller shown in Fig. 1.

Fig. 5 is a block diagram showing a function of a pump tilting controller in a hydraulic drive system according to a second embodiment of the present invention.

Fig. 6 is a block diagram showing a function of a pump tilting controller in a hydraulic drive system according to a third embodiment of the present invention.

Fig. 7 is a diagram of a hydraulic drive system according to a fourth embodiment of the present invention.

Fig. 8 is a block diagram showing a function of a pump tilting controller shown in Fig. 7.

Fig. 9 is a block diagram showing a function of a pump tilting controller in a hydraulic drive system according to a fifth embodiment of the present invention.

Fig. 10 is a block diagram showing a function of a pump tilting controller in a hydraulic drive system according to a sixth embodiment of the present invention.

Fig. 11 is a diagram of a hydraulic drive system according to a seventh embodiment of the present invention.

Fig. 12 is a block diagram showing a function of a pump tilting controller shown in Fig. 11.

BEST MODE FOR CARRYING OUT THE INVENTION

Hereinafter, the present invention will be described in conjunction with illustrated embodiments.

First Embodiment

A first embodiment of the present invention will be explained with reference to Figs. 1 to 4.

In Fig. 1, a hydraulic drive system according to this embodiment comprises a variable displacement hydraulic pump 1 driven by a prime mover (not shown) and having a displacement volume varying mechanism (hereinafter represented by a swash plate), a plurality of hydraulic cylinders or actuators 3A, 3B... (hereinafter represented by 3A, 3B) connected to the hydraulic pump 1 in parallel and driven by a hydraulic fluid delivered from the hydraulic pump 1, a plurality of flow control valves 40A, 40B... (hereinafter represented by 40A, 40B) for respectively controlling flow rates of the hydraulic fluid supplied to the plurality of hydraulic cylinders and controlling driving of these hydraulic cylinders, a plurality of control levers 5A, 5B... (hereinafter represented by 5A, 5B) for instructing respective flow rates to the plurality of flow control valves, input amount sensors 50A, 50B... (hereinafter represented by 50A, 50B) for outputting electric signals proportional to respective input amounts from the control levers, flow rate sensors 10A, 10B... (hereinafter represented by 10A, 10B) for detecting respective flow rates of the hydraulic fluid supplied to the hydraulic cylinders, valve controllers 11A, 11B... (hereinafter represented by 11A, 11B) for respectively controlling driving of the flow control valves 40A, 40b based on signals from the input amount sensors 50A, 50B and the flow rate sensors 10A, 10B, a pump tilting controller 12 for calculating a tilting command value (target displacement volume) of the swash plate of the hy-

draulic pump 1 based on signals from the valve controllers 11A, 11B, and a regulator 20 for driving the swash plate 1a of the hydraulic pump 1 based on a signal from the pump tilting controller 12.

The flow control valves 40A, 40B are of solenoid actuated valves electromagnetically driven with respective control signals from the valve controllers 11A, 11B. As the input amount sensors 50A, 50B, potentiometers are used by which operation of the control levers 5A, 5B in one direction from their neutral positions is given with a "+" sign and their operation in the other direction is given with a "+" sign. The flow rate sensors 10A, 10B can be of, for example, the turbine flow type, the volume type or the Doppler type. The regulator 20 has a solenoid valve operated in response to the signal from the pump tilting controller 12, and the swash plate 1a is driven through operation of that solenoid valve. The valve controllers 11A, 11B and the pump tilting controller 12 each comprise a microcomputer. Alternatively, these controllers may be constituted by one common microcomputer.

The valve controllers 11A, 11B and the pump tilting controller 12 have control functions shown in block diagrams of Figs. 2 to 4. These control functions will be apparent from the following description of operation of this embodiment.

Now, when the control lever 5A, for example, is operated, its input amount is detected by the input amount sensor 50A and applied to the valve controller 11A. As shown in Fig. 2, the valve controller 11A calculates a deviation ΔQ_1 between a detected input amount X_1 and a flow rate Y_1 detected by the flow rate sensor 10A in a subtracter 110, integrates the deviation ΔQ_1 in an integrator 111, and further calculates an opening command value K_1 by multiplying a gain K_i. In this embodiment, taking into account that the flow rate sensor 10A always produces a positive output, an absolute value circuit 114 takes an absolute value of the input amount X_1, the absolute value being compared with the detected flow rate Y_1. A switching control unit 112 outputs a digital value "1" when the sign of the input amount X_1 (i.e., the direction in which the control lever 5A is operated) is "+", and a digital value "0" when it is "-". Thus, the opening command value K_1 is output to one side of the flow control valve 40A in match with the operating direction of the control lever 5A through a switch 113 under control of the switching control unit 112. When the input amount (instructed flow rate) X_1 becomes equal to the detected flow rate (actual flow rate) Y_1, the opening command value K_1 comes into a steady state.

Through the foregoing feedback control, the opening degree of the flow control valve 40A is controlled depending on the input amount from the control lover in such a manner that, even with change in viscosity of the working oil and other factors, the flow control valve 40A is precisely controlled to such an opening as adapted to provide the instructed flow rate. Hereinafter, that control of the flow control valve will be referred to as flow servo control.

Also, when the control lever 5B is operated, the flow servo control is performed by the valve controller 11B in exactly the same manner as mentioned above. When the control lever 5A and the control lever 5B are both operated, the valve controllers 11A, 11B implement the same flow servo control independently of each other. Note that status amounts and calculated values relating to the valve controller 11B are indicated by adding a suffix 2.

Fig. 3 shows a modification in which another function is added to the functions shown in Fig. 2. In Fig. 3, the same components as those in Fig. 2 are denoted by the same reference numerals. Denoted by 116 is a proportional element Kp for the deviation ΔQ used to improve responsivity of the control, and 117 is a differentiation element $Kd \cdot S$ for the deviation ΔQ used to provide stability in the control. The remaining functions are the same as shown in Fig. 2.

In parallel to the foregoing flow servo control by the valve controller 11A, the pump tilting controller 12 makes control as shown in Fig. 4. More specifically, in Fig. 4, the pump tilting controller 12 receives the deviations (hereinafter referred to as flow rate deviations) ΔQ_1, ΔQ_2 calculated by the subtracters 110 of the valve controllers 11A, 11B shown in Fig. 2. Note that the pump tilting controller 12 receives the flow rate deviations ΔQ_1 to ΔQ_n in Fig. 4 on an assumption that the hydraulic actuators, the flow control valves, the valve controllers, etc. are each provided in number of n. The pump tilting controller 12 calculates the total $\Delta\Sigma Q$ of those flow rate deviations ΔQ_1 to ΔQ_n in an adder 120. An output $\Sigma\Delta Q$ of the adder 120 is compared in a subtracter 122 with a reference deviation ΔQ_{ref} which is set as a constant in a deviation setting unit 121 beforehand, thereby calculating a value equal to a result of subtracting the latter from the former. The value obtained by the subtracter 122 is further subjected to calculation in an integrator 123 which has the same function as the integrator 111 shown in Fig. 2, and the calculated result is output as a tilting command value L to the regulator 20. In accordance with the tilting command value L, the regulator 20 controls tilting of the swash plate 1a of the hydraulic pump 1 for controlling the delivery rate of the hydraulic pump 1.

Operation of the pump tilting controller 12 will now be considered. As explained above, the valve controllers 11A, 11B implement the flow servo con-

trol for the flow control valves 40A, 40B so that the deviations ΔQ_1, ΔQ_2 between the instructed flow rates (demanded flow rates) corresponding to the input amounts X_1, X_2 and the detected flow rates (actual flow rates) Y_1, Y_2 each become zero. In contrast, the pump tilting controller 12 controls the delivery rate of the hydraulic pump 1 based on the integrated value of the value resulted by subtracting the reference deviation ΔQ_{ref} from the total $\Sigma\Delta Q$ of the flow rate deviations. This implies that the pump delivery rate is controlled so that the total of the detected flow rates Y_1, Y_2 becomes smaller than the total of the demanded flow rates by a predetermined flow rate corresponding to the reference deviation ΔQ_{ref}. Thus, the delivery rate of the hydraulic pump 1 is controlled to a flow rate smaller than the total demanded flow rate by a predetermined flow rate corresponding to the reference deviation ΔQ_{ref}.

Accordingly, when only the control lever 5A is operated, the hydraulic cylinder 3A is supplied with the hydraulic fluid at a flow rate smaller the reference deviation ΔQ_{ref} than that corresponding to the input amount from the control lever 5A, although the valve controller 11A performs the flow servo control for the flow control valve 40A. Therefore, the opening of the flow control valve 40A is controlled to its maximum value and the resulting smaller pressure loss by the flow control valve 40A makes it possible to suppress the delivery pressure of the hydraulic pump 1 at a lower level. A reduction in the supply flow rate by the amount of ΔQ_{ref} will not give rise to any trouble in practical use if the reference deviation ΔQ_{ref} is set to a value as small as possible while achieving the intended function.

While the above explanation is concerned with the case of driving the hydraulic actuator 3A only, it similarly applies to the case of simultaneously driving the plural hydraulic actuators. More specifically, those hydraulic actuators other than that producing the maximum load pressure are supplied with the hydraulic fluid at respective demanded flow rates through the flow servo control by the associated valve controllers, but the hydraulic actuator producing the maximum load pressure is supplied with the hydraulic fluid at a flow rate smaller than the reference deviation ΔQ_{ref} than the demanded flow rate and the associated flow control valve is maximized in its opening through the flow servo control.

From the standpoint of saving in energy, the delivery pressure of the hydraulic pump is desirably the same as maximum one of load pressures produced by the plural hydraulic actuators. However, since the hydraulic fluid is supplied via the flow control valve to the hydraulic actuator producing the maximum load pressure, it is inevitable that the delivery pressure of the hydraulic pump is raised by an amount of the pressure loss produced by the flow control valve. Conversely, this means that by making the above pressure loss smaller, the delivery pressure of the hydraulic pump can be ideally suppressed to a necessary lowest value. In this embodiment, because the flow control valve associated with the hydraulic actuator producing the maximum load pressure is maximized in its opening, as mentioned above, the pressure loss produced by the flow control valve is minimized, enabling the delivery pressure of the hydraulic pump to be ideally suppressed to a necessary lowest value.

Also, the fact that the delivery rate of the hydraulic pump 1 is controlled to a value smaller the reference deviation ΔQ_{ref} than the demanded flow rate has an important meaning below in this embodiment.

Let it be supposed that the reference deviation ΔQ_{ref} is not set in this embodiment. This corresponds to the case that the hydraulic drive system shown in Fig. 1 has the pump tilting controller not provided with the components 121, 122 in the block diagram of Fig. 4. Let it be also supposed that the delivery rate of the hydraulic pump happens to become larger than the demanded flow rate in the above arrangement. This condition may occur, for example, if the flow servo control functions, prior to a reduction in the delivery rate of the hydraulic pump, for restricting the opening of the flow control valve to achieve the target flow rate, when the input amount from the control lever is reduced. In such a case, the surplus hydraulic fluid is returned to a reservoir via a relief valve provided, though not shown in Fig. 1, near a pump delivery port for the safety purpose. Thus, the pump delivery pressure is raised up to a set pressure of the relief valve no matter how light the actuator load may be. At this time, because of being kept under the flow servo control by the valve controllers 11A, 11B, the flow control valves are controlled such that their openings are reduced to supply the hydraulic fluids at respective predetermined flow rates even with the associated actuators having light loads. Accordingly, the total flow rate deviation $\Sigma\Delta Q$ becomes 0 and the output of the integrator 123 is not changed, meaning that the pump tilting amount remains the same and the above relief condition is maintained in such as case. In other words, the hydraulic pump cannot generate the required flow rate and pressure only, making the system fail to function as a practical one.

In contrast, with this embodiment, even if the system comes into the relief condition and the total flow rate deviation $\Sigma\Delta Q$ becomes 0, the tilting amount of the hydraulic pump is gradually reduced with the presence of ΔQ_{ref},enabling the system to escape from the relief condition. As a result, the

hydraulic pump can be efficiently operated while generating the required flow rate and pressure only. Thus, the presence of the reference deviation ΔQ_{ref} makes it first possible to, in parallel to the flow servo control, implement control of the pump delivery rate in accordance with the demanded flow rate.

Furthermore, this embodiment uses the total flow rate deviation $\Sigma\Delta Q$, rather the input amounts X_1, X_2 from the control levers, for controlling the pump delivery rate in accordance with the demanded flow rate, and this feature provides the following important action.

Consider first the case that the delivery rate of the hydraulic pump is controlled by receiving the input amounts X_1, X_2 from the control levers without introducing the reference deviation ΔQ_{ref}. In this case, if there exist no errors in the flow rate sensors 10A, 10B, the regulator 20 and so forth, no problems occur. Stated otherwise, if so, the pump delivery rate can be controlled to be coincident with the demanded flow rate in parallel to the flow servo control. Generally, however, the sensors contain errors in terms of detection accuracy. Accordingly, it is supposed that while the total of the input amounts X_1, X_2 from the control levers is recognized as 100 ℓ/min, for example, and the hydraulic pump actually delivers the hydraulic fluid at a flow rate of 100 ℓ/min, the hydraulic fluid is supplied to the actuators only at an actual flow rate of 99 ℓ/min in a steady state for the flow control valves are subjected to the flow servo control independently of each other. This case happens, for example, if one flow rate sensor detects a flow rate of 51 ℓ/min despite the actual flow rate being 50 ℓ/min. In such a case, the hydraulic fluid is delivered from the hydraulic pump at 100 ℓ/min, whereas the actuators are supplied with only at 99 ℓ/min, resulting in the problem that there occurs a surplus flow rate of 1 ℓ/min which is released similarly to the above-mentioned case. Accordingly, the hydraulic pump requires power greater than necessary and efficiency of the entire system is lowered.

A first method for avoiding the above drawback is to set the pump delivery rate at a relatively small value such that the delivery rate of the hydraulic pump becomes still insufficient or smaller than the value obtained by subtracting accumulated all errors possibly occurred in the sensors, the regulator and so forth from the required pump delivery rate. This can be realized by providing a reference deviation ΔQ_{ref} as with this embodiment. Note that the first method will be described in detail later as another embodiment (see Figs. 11 and 12). In that case, the reference deviation ΔQ_{ref} is given by approximately 1 to 5 % of the maximum delivery rate of the hydraulic pump x N (where N is the number of hydraulic actuators). Assuming now that accuracy of the flow rate sensors 10A, 10B are each ± 2 ℓ/min, there are three hydraulic actuators, and delivery rate accuracy of the hydraulic pump is 3 ℓ/min, by way of example, the reference deviation must be set as follows:

$$\Delta Q_{ref} \geqq 2\ (\ell/\text{min}) \times 3 + 3\ (\ell/\text{min}) = 9\ (\ell/\text{min})$$

A second method for avoiding the above drawback is to use the total flow rate deviation $\Sigma\Delta Q$ as practiced in this embodiment. More specifically, using the total flow rate deviation $\Sigma\Delta Q$ is equivalent to inform the hydraulic pump of whether the flow rates are sufficient or deficient, based on the result of the flow servo control on the hydraulic actuator side and, therefore, the aforesaid relief condition will not occur due to accuracy of the flow rate sensors 10A, 10B. Also, since the tilting amount of the hydraulic pump is only increased and decreased based on information about sufficiency or deficiency in the flow rates from the hydraulic actuator side by using the integrator 123 rather than specifying an absolute value of the tilting amount, accuracy on the pump control side will never be affected.

However, in the case of using the total flow rate deviation $\Sigma\Delta Q$, the relief condition may occur for another reason as mentioned above in the absence of the reference deviation ΔQ_{ref}, making the system fail to function as a practical one. Because ΔQ_{ref} used in this case is not affected by accuracy of the sensors and the pump control side, it can be set to a very small value in consideration of, strictly speaking, an error possibly occurred in calculation by the controllers which generally comprise microcomputers. The reference deviation ΔQ_{ref} is approximately 0.1 to 3 % of the maximum delivery rate of the hydraulic pump. Accordingly, it is possible to minimize a lack of the flow rate for the hydraulic actuator producing the maximum load pressure and to achieve the accurate flow control. It should be understood that for a response becomes slow in the transient region if the reference deviation ΔQ_{ref} is too small, the reference deviation ΔQ_{ref} is actually determined, taking into account responsivity as well.

With this embodiment, as explained above, since the flow servo control is performed so as to make the opening of the flow control valve in match with the demanded flow rate, the hydraulic actuator driven through the flow control valve can be operated with high accuracy without being affected by oil temperatures, etc. Also, since the flow control valve associated with the hydraulic actuator producing the maximum load pressure is maximized in its opening, the pressure loss can be suppressed to a small value.

Further, with this embodiment, since the delivery rate of the hydraulic pump is controlled by using the total flow rate deviation $\Sigma\Delta Q$, the pump delivery rate can be controlled by setting a small value of the reference deviation ΔQ_{ref} without causing the relief condition, and an influence of the reference deviation upon the flow control is minimized to enable the accurate flow control.

Second Embodiment

A second embodiment of the present invention will be described with reference to Fig. 5. In this embodiment, a pump tilting controller 12A has functions different from those shown in Fig. 4 only in that a maximum value selector 124 is provided instead of the adder 120, the remaining functions are the same. The maximum value selector 124 selects maximum one of the deviations ΔQ_1, $\Delta Q_2 ... \Delta Q_n$ and outputs it to the subtracter 122. Selecting the maximum flow rate deviation by the maximum value selector 124 in this embodiment implies that tilting control of the hydraulic pump is performed by using information about the actuator of which flow rate is most insufficient, whereby a transient response is improved.

Referring back to Fig. 1, when the hydraulic cylinder 3A is driven by operating only the control lever 5A, the valve controller 11A implements the flow servo control for the flow control valve 40A in such a manner as explained above. In the case of sole operation of one hydraulic actuator, because the total flow rate deviation $\Sigma\Delta Q$ and the maximum flow rate deviation have the same value, the pump tilting controller 12A implements the control with the same functions as those of the first embodiment shown in Fig. 4. Specifically, the flow rate deviation ΔQ_1 as a deviation between the input amount X_1 and the detected flow rate Y_1 is selected as the maximum flow rate deviation by the maximum value selector 124, and the pump delivery rate is controlled to become smaller the reference deviation ΔQ_{ref} than the demanded flow rate. Also, the flow control valve 40A is controlled to have its maximum opening.

Let it be supposed that, under the above condition, the control lever 5B is operated to drive the hydraulic cylinder 3B and the hydraulic cylinder 3B produces a higher load pressure than the hydraulic cylinder 3A. In this case, the delivery pressure of the hydraulic pump 1 is raised and, at the same time, the tilting amount of the swash plate 1a of the hydraulic pump 1 must be increased, thereby giving rise to a transient phenomenon below.

For the flow control valve 40A, since the pressure is raised in a maximum opening condition, the flow rate becomes too large and the flow rate deviation ΔQ_1 takes a negative value. On the other hand, for the flow control valve 40B, since the pressure is raised in a maximum opening condition, the flow rate becomes insufficient until the tilting amount of the hydraulic pump 1 is increased, and the flow rate deviation ΔQ_2 takes a positive value.

In such a condition, $(\Delta Q_2 - |\Delta Q_1|) - \Delta Q_{ref}$ is applied to the integrator 123 in the first embodiment having the functions shown in Fig. 4. Meanwhile, ΔQ_2 is selected by the maximum value selector 124 and $\Delta Q_2 - \Delta Q_{ref}$ is applied to the integrator 123 in the this embodiment having the functions shown in Fig. 5. Thus, the value (absolute value) applied to the integrator 123 is larger in this embodiment of Fig. 5 than in the first embodiment of Fig. 4. Accordingly, the tilting command value L can be increased at a higher speed and responsivity of the tilting in the transient region can be improved.

In a steady state, the flow rate supplied to only the hydraulic cylinder 3B as the hydraulic actuator producing the maximum load pressure becomes insufficient by an amount of the reference deviation ΔQ_{ref} and the flow control valve 40B is controlled to be maximized in its opening. Also, the flow rate deviation ΔQ_2 (= + ΔQ_{ref}) for the hydraulic cylinder 3B is selected as the maximum flow rate deviation by the maximum value selector 124 and the input to the integrator 123 becomes 0, thereby keeping the pump tilting amount constant. At this time, because of the flow rate deviation ΔQ_1 for hydraulic cylinder 3A being 0, there is obtained the same result as the case that the total flow rate deviation $\Sigma\Delta Q$ is calculated and output the integrator 123 in the first embodiment having the functions shown in Fig. 4. In other words, the maximum value selector 124 functions as means for calculating the total flow rate deviation $\Sigma\Delta Q$ in a steady state.

As a result, with this embodiment, it is possible to not only obtain the same advantage as that of the first embodiment, but also achieve the pump tilting control with a good response since the tilting control of the hydraulic pump is performed by using the maximum flow rate deviation as information about the actuator of which flow rate is most insufficient.

Third Embodiment

A third embodiment of the present invention will be described with reference to Fig. 6. In the foregoing embodiments, the reference deviation ΔQ_{ref} has been described as a preset constant. It has also been stated that the satisfactory operation can be achieved by setting the reference deviation ΔQ_{ref} to be approximately 0.1 to 3 % of the maximum delivery rate of the hydraulic pump in consid-

eration of responsivity in the transient region. However, because the hydraulic actuator operated under the maximum load pressure is always supplied with the hydraulic fluid only at a flow rate smaller the deviation ΔQ_{ref} than the demanded flow rate, the deviation ΔQ_{ref} is desirably made as small as practicable in fine operation requiring higher accuracy. This embodiment includes a function to meet such a requirement.

In Fig. 6, a pump tilting controller 12B receives, in addition to the signals of the flow rate deviations ΔQ_1, $\Delta Q_2 ... \Delta Q_n$ from the valve controllers 11A, 11B, the signals of absolute values of the input amounts X_1, $X_2 ... X_n$ from the control levers and calculates the tilting command value L based on these signals. Specifically, the pump tilting controller 12B has an adder 126 for adding the absolute values of the input amounts X_1, $X_2 ... X_n$, and a multiplier 127 for multiplying the total of these absolute values of the input amounts by a constant Kx. An output of the multiplier 127 becomes the deviation ΔQ_{ref}. The remaining functions are the same as those shown in Fig. 4.

With this embodiment thus arranged, the total of the demanded flow rates is calculated by the adder 126 and the deviation ΔQ_{ref} is determined by multiplying the total demanded flow rate by the proper constant Kx. Thus, the deviation ΔQ_{ref} is determined in proportion to the total demanded flow rate, with the result of that particularly when the total demanded flow rate is small, a control error in the flow rate supplied to the hydraulic actuator producing the maximum load pressure can be made smaller. On the contrary, when the total demanded flow rate is large, the deviation ΔQ_{ref} also becomes large to permit the control with a good response in the transient region.

Fourth Embodiment

A fourth embodiment of the present invention will be described with reference to Figs. 7 and 8. This embodiment is intended to provide another method of determining the reference deviation ΔQ_{ref}. In Fig. 7, the same components as those in Fig. 1 are denoted by the same reference numerals.

In Fig. 7, a hydraulic drive system of this embodiment includes shuttle valves 13A, 13B... (hereinafter represented by 13A, 13B), pressure sensors 14A, 14B... (hereinafter represented by 14A, 14B), and a maximum load pressure selector 15. The pressure sensors 14A, 14B respectively output, through the shuttle valves 13A, 13B, electric signals V_1, V_2 proportional to load pressures of the hydraulic cylinders 3A, 3B. The maximum load pressure selector 15 receives the signals from the pressure sensors 14A, 14B and outputs a signal N corresponding to the hydraulic actuator which produces a maximum load pressure. A pump tilting controller 12C has the same functions as those of the pump tilting controller 12 shown in Fig. 1 except for its part.

Fig. 8 is a block diagram for explaining functions of the pump tilting controller 12C. The pump tilting controller 12C receives, in addition to the signals of the flow rate deviations ΔQ_1, $\Delta Q_2 ... \Delta Q_n$ from the valve controllers 11A, 11B, the signals of absolute values of the input amounts X_1, $X_2 ... X_n$ from the control levers and the signal N from the maximum load pressure selector 15. The pump tilting controller 12C has a switching unit 129 for receiving the absolute values of the input amounts X_1, $X_2 ... X_n$ and the signal N from the maximum load pressure selector 15 and selecting the absolute value of the input amount corresponding to the hydraulic actuator which produces the maximum load pressure, and a multiplier 127 for multiplying the selected absolute values of the input amount by a constant Kx. An output of the multiplier 127 becomes the deviation ΔQ_{ref}. The remaining functions are the same as those shown in Fig. 4.

In this embodiment, as mentioned before, the hydraulic actuator producing the maximum load pressure is always supplied with the hydraulic fluid at a flow rate smaller the reference deviation ΔQ_{ref} than the demanded flow rate. Therefore, by changing the reference deviation ΔQ_{ref} depending on the instructed flow rate for that hydraulic actuator, control accuracy can be further increased. The pressure sensors 14A, 14B and the maximum load pressure selector 15 shown in Fig. 7 are provided for the above purpose. More specifically, the maximum load pressure selector 15 functions as means for detecting the hydraulic actuator producing the maximum load pressure; i.e., it selects the hydraulic actuator producing the maximum load pressure based on the pressure signals applied thereto and outputs the signal N corresponding to that hydraulic actuator. The pump tilting controller 12C receives the signal N at the switching unit 129, selects one of the absolute values of the input amounts from the control levers corresponding to that hydraulic actuator, and outputs it to the multiplier 127. As a result, the hydraulic actuator producing the maximum load pressure is surely supplied with the hydraulic fluid at a flow rate smaller than the demanded flow rate by a value equal to the product of the demanded flow rate and the constant Kx. Given the value Kx being 0.01, by way of example, the deviation ΔQ_{ref} is 1 % of the instructed flow rate for the hydraulic actuator.

With this embodiment, since the reference deviation is determined depending on the demanded flow rate for the hydraulic actuator producing the

maximum load pressure, a control error in the flow rate supplied to that hydraulic actuator can be made smaller when the demanded flow rate is small. On the contrary, when the demanded flow rate is large, the deviation ΔQ_{ref} also becomes large to permit the control with a good response in the transient region.

Fifth Embodiment

A fifth embodiment of the present invention will be described with reference to Fig. 9. While the above fourth embodiment uses the maximum load pressure selector as means for detecting the hydraulic actuator producing the maximum load pressure, this embodiment adopts another method in this respect.

In Fig. 9, a pump tilting controller 12D of this embodiment has a maximum value selector 13 which receives the opening command values K_1, K_2...K_n calculated by the respective valve controllers, selects the hydraulic actuator corresponding to the maximum opening command value as the hydraulic actuator producing the maximum load pressure, and then outputs the corresponding signal N. Since the hydraulic actuator producing the maximum load pressure is controlled with the maximum opening, the hydraulic actuator producing the maximum load pressure can be also detected in this embodiment by selecting the hydraulic actuator corresponding to the maximum opening command value. In response to the signal N from the maximum value selector 130, the switching unit 129 selects one of the absolute values of the input amounts from the control levers corresponding to that hydraulic actuator, and outputs it to the multiplier 127. The remaining functions are the same as those shown in Fig. 4.

This embodiment can also provides the similar advantage to the fourth embodiment shown in Figs. 7 and 8.

Sixth Embodiment

A sixth embodiment of the present invention will be described with reference to Fig. 10. This embodiment is intended to improve responsivity of the pump tilting control.

In Fig. 10, a pump tilting controller 12E receives the signals of the flow rate deviations ΔQ_1, ΔQ_2..ΔQ_n from the valve controllers 11A, 11B and the signals of absolute values of the input amounts X_1, X_2...X_n from the control levers, and calculates the tilting command value L based on these signals. Specifically, the pump tilting controller 12E has an adder 131 for adding the absolute values of the input amounts X_1, X_2...X_n, a multiplier 132 for multiplying the total of these absolute values of the input amounts by a constant Ky, and an adder 133 for adding an output of the multiplier 132 to the output of the integrator 123. An output of the multiplier 132 is used as a modification value for the tilting command value and an output of the adder 133 becomes the final tilting command value L. The remaining functions are the same as those shown in Fig. 4.

With this embodiment thus arranged, since the modification value proportional to the total of the absolute values of the input amounts X_1, X_2...X_n is added in the adder 133 to the tilting command value obtained as an integrated value, there can be provided an advantage of improving responsivity in the transient region. Note that for the same reason as stated in connection with the second embodiment of Fig. 5, a maximum value selector may be used instead of the adder 131.

Seventh Embodiment

A seventh embodiment of the present invention will be described with reference to Figs. 11 and 12. In this embodiment, the delivery rate of the hydraulic pump is controlled in accordance with the demanded flow rate by using the total of the input amounts from the control levers rather than the total $\Sigma\Delta Q$ of the flow rate deviations.

In Fig. 11, a hydraulic drive system of this embodiment includes a pump tilting controller 12F for receiving the signals of the input amounts X_1, X_2 from the control levers 5A, 5B detected by the input amount sensors 50A, 50B, and calculating the tilting command value.

In the pump tilting controller 12F, as shown in Fig. 12, absolute values of the input amounts X_1, X_2 from the control levers 5A, 5B in an absolute value circuit 140 and these absolute values are added in an adder 141 to determine the total ΣX of the input amounts. An output ΣX of the adder 141 is compared in a subtracter 142 with a reference deviation X_{ref} set as a constant in a deviation setting unit 143 beforehand, thereby calculating a value equal to a result of subtracting the latter from the former. The value obtained by the subtracter 142 is further subjected to calculation in a proportion unit 144 and the calculated result is output as a tilting command value L to the regulator 20. In accordance with the tilting command value L, the regulator 20 controls tilting of the swash plate 1a of the hydraulic pump 1 for controlling the delivery rate of the hydraulic pump 1.

As stated before, when the delivery rate of the hydraulic pump is controlled by using the total ΣX of the input amounts from the control levers without introducing the reference deviation X_{ref}, the delivery rate of the hydraulic pump may become larger than the flow rate actually passing through the flow

control valve due to errors in the flow rate sensors 10A, 10B, the regulator 20 and so forth, which results in the problem that the surplus flow rate may be released. Setting of the reference deviation X_{ref} makes it possible to eliminate that problem and achieve economical operation. In this embodiment, the reference deviation X_{ref} is given by approximately 1 to 5 % of the maximum delivery rate of the hydraulic pump x N (where N is the number of hydraulic actuators).

Further, as with the case of using the total flow rate deviation $\Sigma\Delta Q$, since the pump delivery rate is kept smaller than the demanded flow rate, the flow control valve associated with the hydraulic actuator producing the maximum load pressure is controlled to be maximized in its opening, whereby the pressure loss can be suppressed to a small value.

Additionally, with this embodiment, since the pump tilting is controlled through an open loop independently of the flow servo control for the valve controllers 11A, 11B, it is possible to ensure stable delivery rate control of the hydraulic pump without causing hunting.

INDUSTRIAL APPLICABILITY

According to the present invention, as described above, since the flow servo control is performed so as to make the opening of the flow control valve in match with the demanded flow rate, the hydraulic actuator driven through the flow control valve can be operated with high accuracy without being affected by oil temperatures, etc. Also, since the flow control valve associated with the hydraulic actuator producing the maximum load pressure is maximized in its opening, the pressure loss can be suppressed to a small value. Further, in the case that the delivery rate of the hydraulic pump is controlled by using the total flow rate deviation $\Sigma\Delta Q$, the pump delivery rate can be controlled by setting a small value of the reference deviation ΔQ_{ref} without causing the relief condition. In addition, accurate flow control can be enabled. Alternatively, in the case that the delivery rate of the hydraulic pump is controlled by using the total input amount ΣX, the pump delivery rate can be controlled not only in a reliable manner without causing the relief condition, but also in a stable manner without causing hunting.

Claims

1. A hydraulic drive system comprising a variable displacement hydraulic pump (1), a plurality of hydraulic actuators (3A, 3B) connected to said hydraulic pump in parallel, a plurality of flow control valves (40A, 40B) for respectively driving said plurality of hydraulic actuators, and a plurality of flow rate instructing means (5A, 5B) for instructing respective flow rates to said plurality of flow control valves, said system further comprising:

   a plurality of flow rate sensor means (10A, 10B) for detecting respective flow rates supplied to said plurality of hydraulic actuators (3A, 3B),

   first control means (11A, 11B) for respectively controlling said plurality of flow control valves (40A, 40B) so that the flow rates detected by said plurality of flow rate sensor means are coincident with the flow rates instructed by said plurality of flow rate instructing means (5A, 5B), and

   second control means (12; 12A - 12F) for controlling a delivery rate of said hydraulic pump (1) such that the delivery rate of said hydraulic pump is smaller by a predetermined flow rate (ΔQ_{ref}; X_{ref}) than the total of the flow rates instructed by said plurality of flow rate instructing means.

2. A hydraulic drive system according to claim 1, wherein said second control means (12; 12A - 12E) controls a displacement volume of said hydraulic pump (1) such that the total of the flow rates detected by said plurality of flow rate sensor means (10A, 10B) is smaller by said predetermined flow rate (ΔQ_{ref}) than the total of the flow rates instructed by said plurality of flow rate instructing means (5A, 5B).

3. A hydraulic drive system according to claim 1, wherein said second control means (12; 12A - 12E) controls the delivery rate of said hydraulic pump (1) by using flow rate deviations (ΔQ_1, ΔQ_2) = resulted from respectively subtracting the flow rates detected by said plurality of flow rate sensor means (10A, 10B) from the flow rates instructed by said plurality of flow rate instructing means (5A, 5B).

4. A hydraulic drive system according to claim 1, wherein said second control means (12; 12A - 12E) comprises first calculation means (120; 124) for calculating the total ($\Sigma\Delta Q$) of flow rate deviations (ΔQ_1, ΔQ_2) resulted from respectively subtracting the flow rates detected by said plurality of flow rate sensor means (10A, 10B) from the flow rates instructed by said plurality of flow rate instructing means (5A, 5B), deviation output means (121; 127) for outputting a value corresponding to said predetermined flow rate as a reference deviation (ΔQ_{ref}), second calculation means (122) for calculating a difference between the total ($\Sigma\Delta Q$) of the flow rate deviations obtained by said

first calculation means and the reference deviation (ΔQ_{ref}) output from said deviation output means, and third calculation means (123) for determining a target displacement volume of said hydraulic pump based on the difference obtained by said second calculation means.

5. A hydraulic drive system according to claim 4, wherein said first calculation means comprises means (120) for adding said flow rate deviations (ΔQ_1, ΔQ_2).

6. A hydraulic drive system according to claim 4, wherein said first calculation means comprises means (124) for selecting a maximum value of said flow rate deviations (ΔQ_1, ΔQ_2).

7. A hydraulic drive system according to claim 1, wherein said second control means (12F) comprises first calculation means (141) for calculating the total (ΣX) of the flow rates instructed by said plurality of flow rate instructing means (5A, 5B), deviation output means (143) for outputting a value corresponding to said predetermined flow rate as a reference deviation (X_{ref}), second calculation means (142) for calculating a difference between the total (ΣX) of the instructed flow rates obtained by said first calculation means and the reference deviation (X_{ref}) output from said deviation output means, and third calculation means (144) for determining a target displacement volume of said hydraulic pump based on the difference obtained by said second calculation means.

8. A hydraulic drive system according to claim 1, wherein said second control means includes deviation output means (121; 127) for outputting a value corresponding to said predetermined flow rate as a reference deviation (ΔQ_{ref}).

9. A hydraulic drive system according to claim 8, wherein said deviation output means (121) stores said reference deviation (ΔQ_{ref}) as a constant beforehand.

10. A hydraulic drive system according to claim 8, wherein said deviation output means includes means (126; 127) for determining said reference deviation (ΔQ_{ref}) depending on the total of the flow rates instructed by said plurality of flow rate instructing means (5A, 5B).

11. A hydraulic drive system according to claim 8, wherein said deviation output means includes means (15; 130) for determining one of said plurality of hydraulic actuators (3A, 3B) which is subjected to a maximum load pressure, means (129) for selecting one of the flow rates instructed by said flow rate instructing means (5A, 5B) which corresponds to said hydraulic actuator subjected to the maximum load pressure, and means (127) for determining said reference deviation (ΔQ_{ref}) depending on said selected instructed flow rate.

12. A hydraulic drive system according to claim 1, wherein said second control means comprises integration means (123) for calculating a target displacement volume of said hydraulic pump adapted to make the delivery rate of said hydraulic pump smaller by said predetermined flow rate (ΔQ_{ref}) than the total of the flow rates instructed by said plurality of flow rate instructing means (5A, 5B), means (131) for calculating the total of the flow rates instructed by said plurality of flow rate instructing means, means (132) for calculating a modification value for said target displacement volume based on the total of said instructed flow rates, and means (133) for adding said modification value to the target displacement volume calculated by said integration means and calculating a final target displacement volume.

FIG. 1

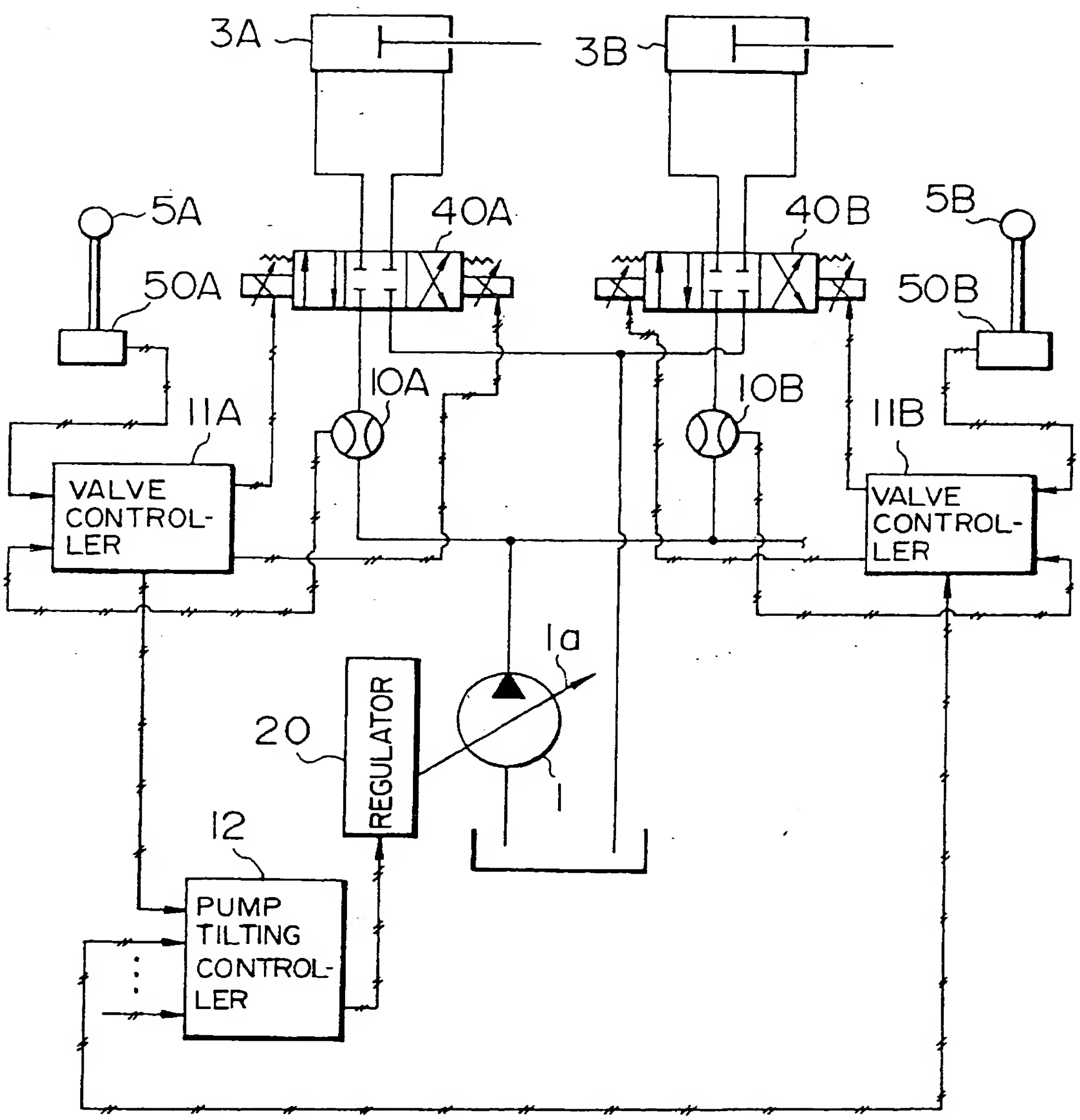

3A
3B
5A
5B
40A
40B
50A
50B
10A
10B
11A
11B
VALVE CONTROL-LER
VALVE CONTROL-LER
1a
20
REGULATOR
1
12
PUMP TILTING CONTROL-LER

FIG.2

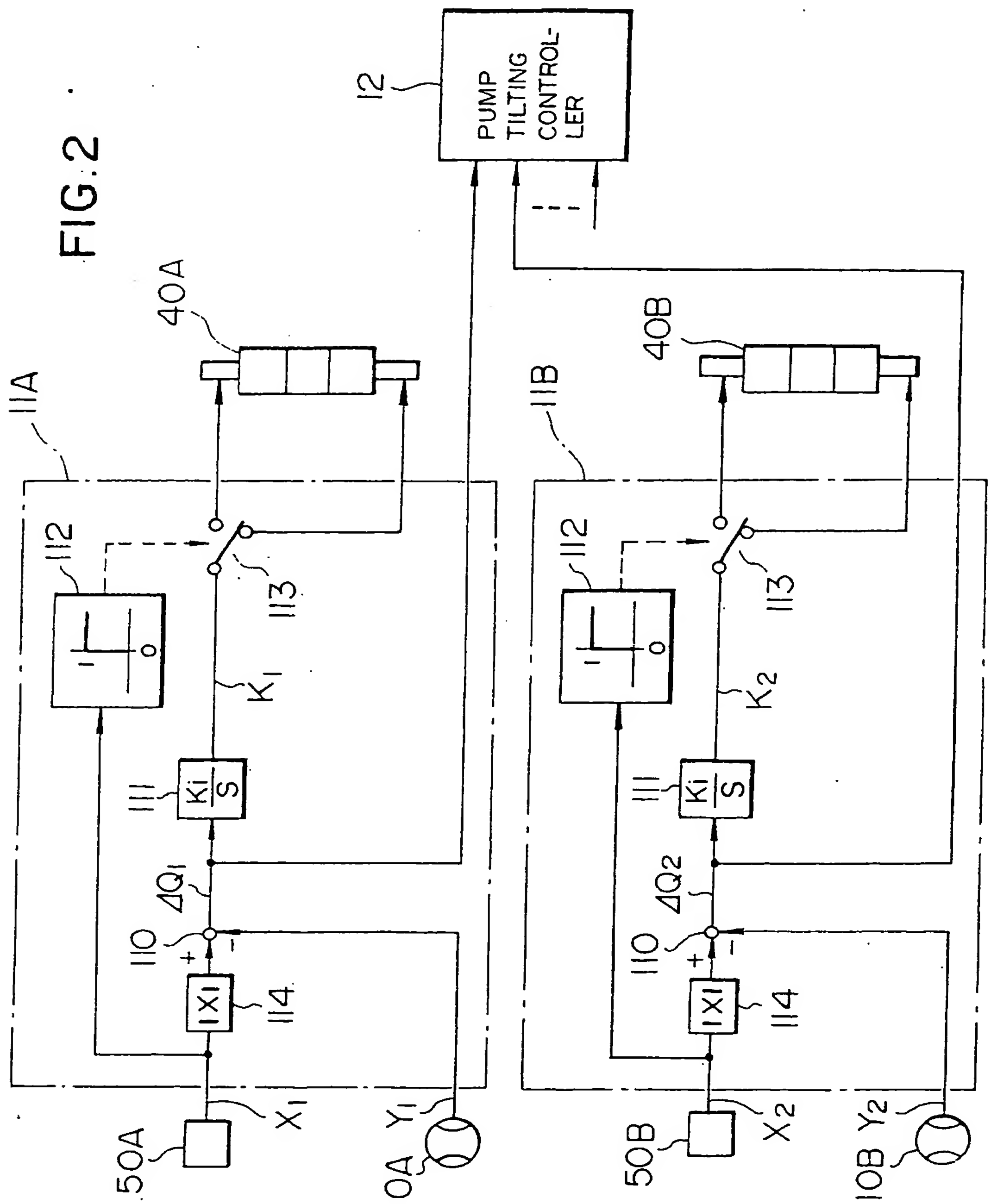
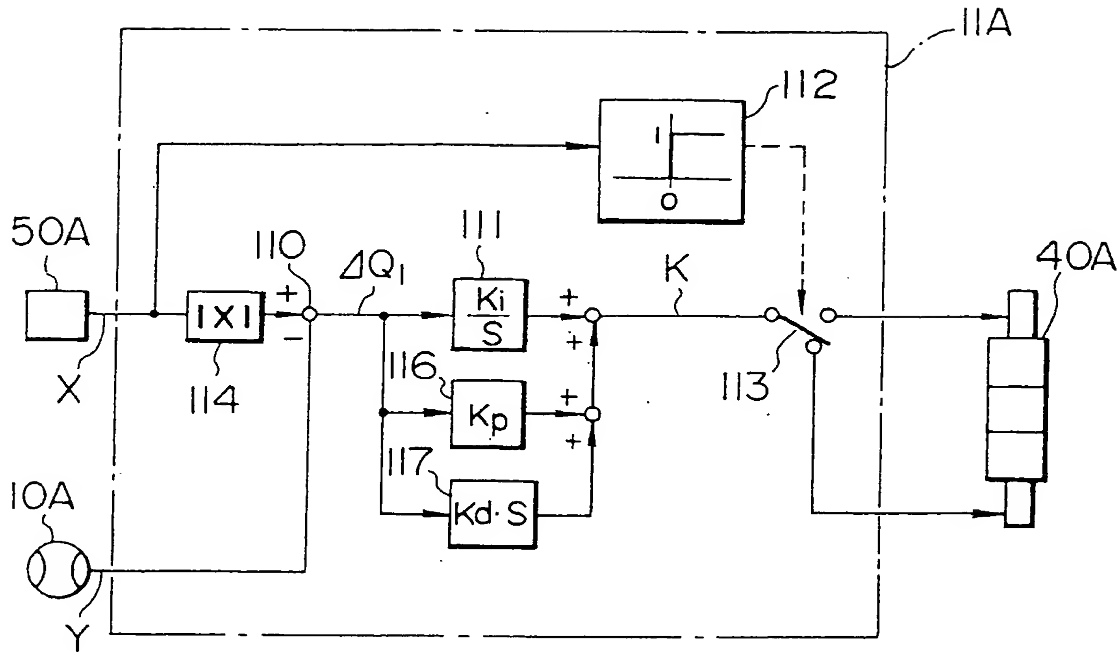

12
PUMP TILTING CONTROLLER
11A
40A
112
113
K1
111
Ki/S
ΔQ1
110
114
|X|
50A
X1
10A
Y1
11B
40B
K2
ΔQ2
50B
X2
10B
Y2

FIG.3


11A
112
1
0
50A
110
ΔQ1
111
Ki
S
K
40A
|X|
X
114
116
Kp
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113
10A
Y

FIG.4

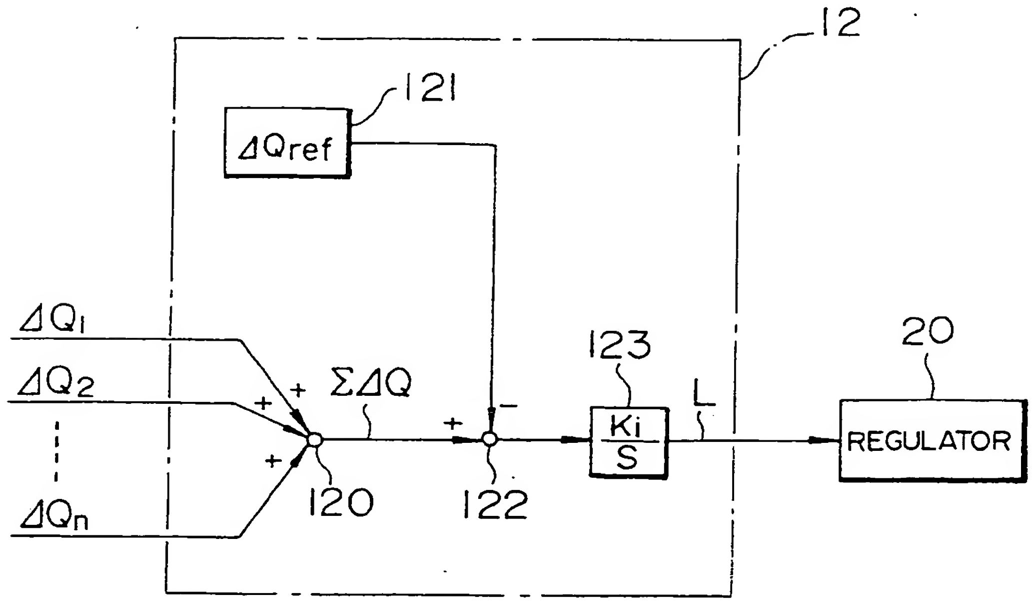

12
121
ΔQref
ΔQ1
ΔQ2
ΔQn
ΣΔQ
123
Ki
S
L
20
REGULATOR
120
122

FIG.5

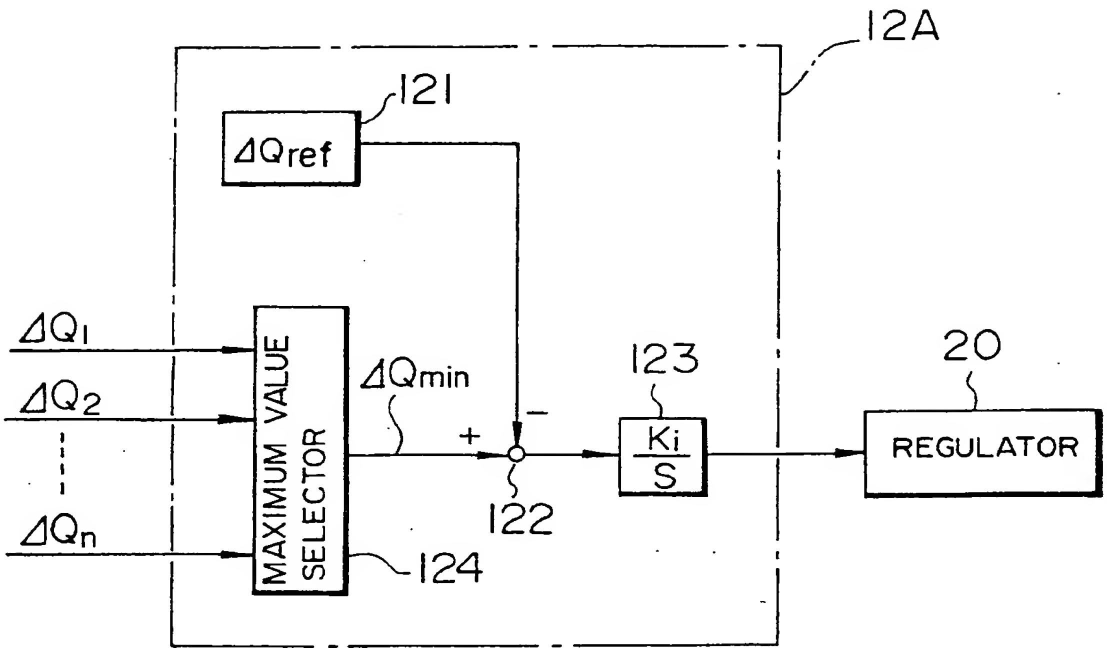

12A
121
ΔQref
ΔQ1
ΔQ2
ΔQn
MAXIMUM VALUE SELECTOR
ΔQmin
+
−
122
123
Ki/S
124
20
REGULATOR

FIG.6

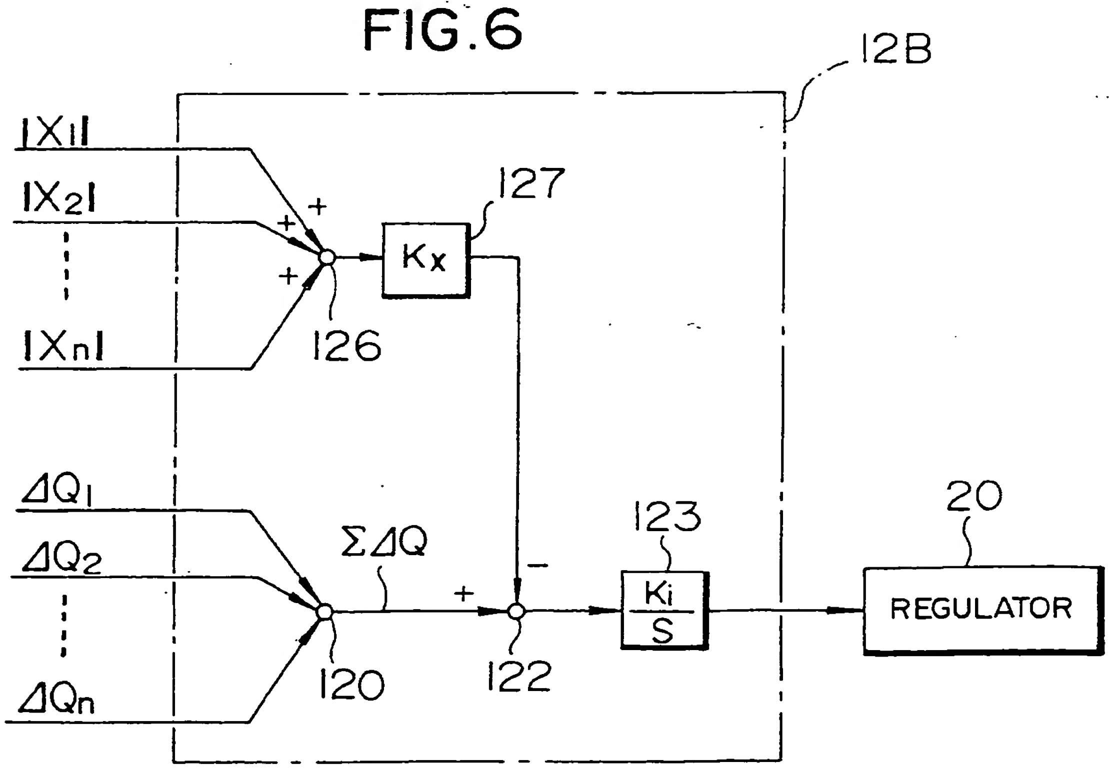

12B
|X1|
|X2|
|Xn|
+
+
+
126
127
Kx
ΔQ1
ΔQ2
ΔQn
ΣΔQ
+
−
120
122
123
Ki/S
20
REGULATOR

FIG. 7

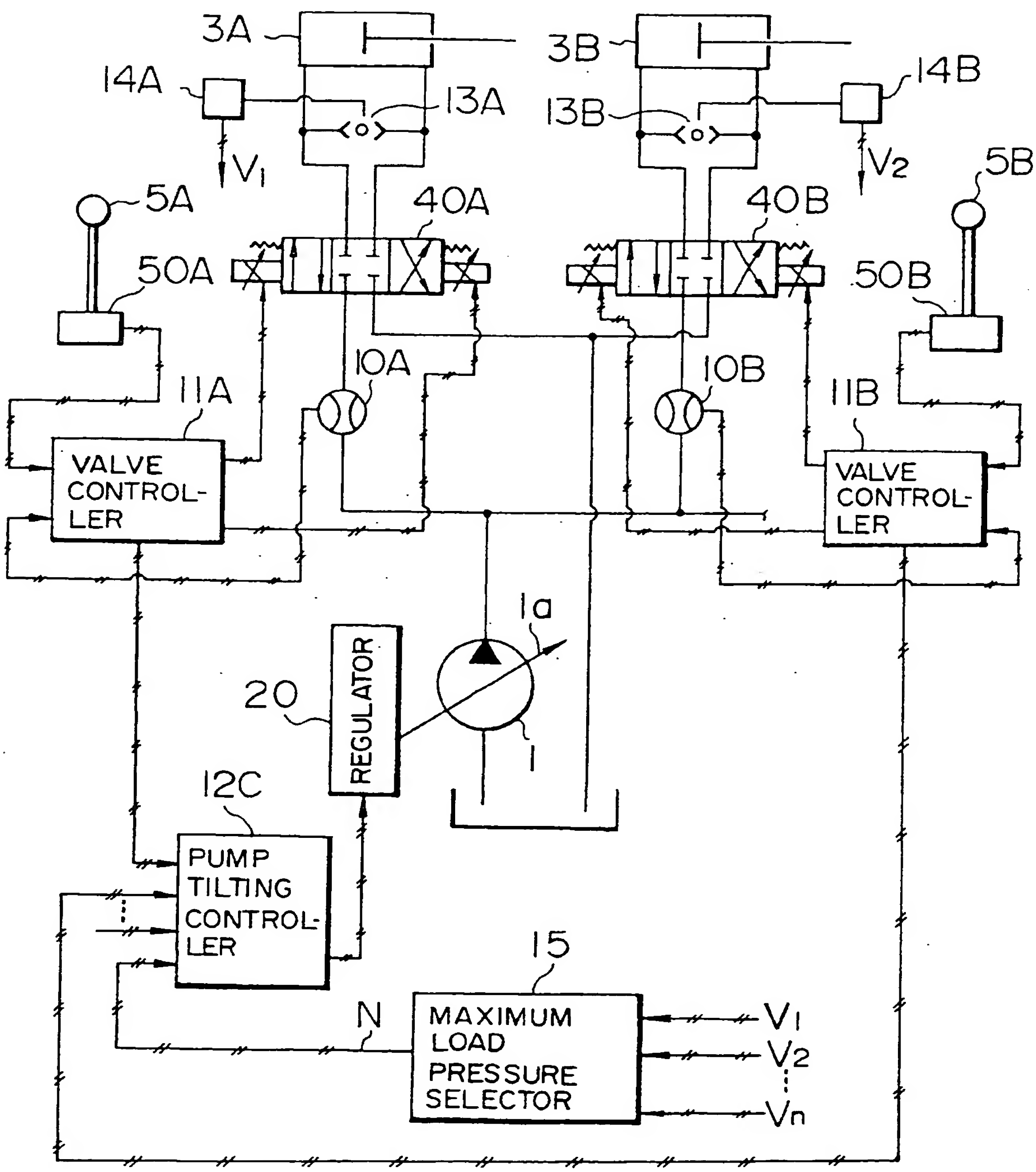

3A
3B
14A
14B
13A
13B
V1
V2
5A
5B
40A
40B
50A
50B
10A
10B
11A
11B
VALVE CONTROL-LER
VALVE CONTROL-LER
1a
20
REGULATOR
1
12C
PUMP TILTING CONTROL-LER
15
N
MAXIMUM LOAD PRESSURE SELECTOR
V1
V2
Vn

FIG. 8

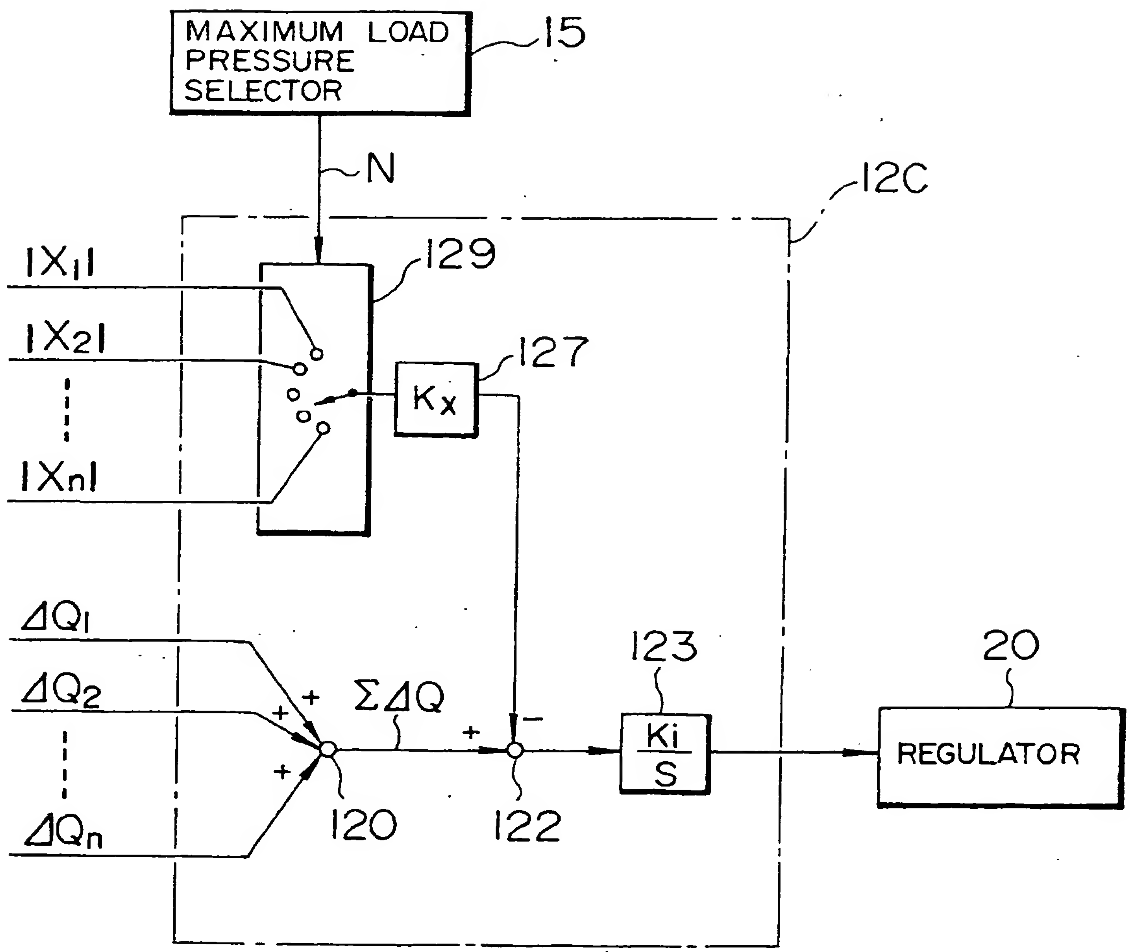

MAXIMUM LOAD PRESSURE SELECTOR
15
N
12C
129
127
K_x
$|X_1|$
$|X_2|$
$|X_n|$
ΔQ_1
ΔQ_2
ΔQ_n
+
+
+
$\Sigma\Delta Q$
+
−
123
$\frac{Ki}{S}$
20
REGULATOR
120
122

FIG. 9

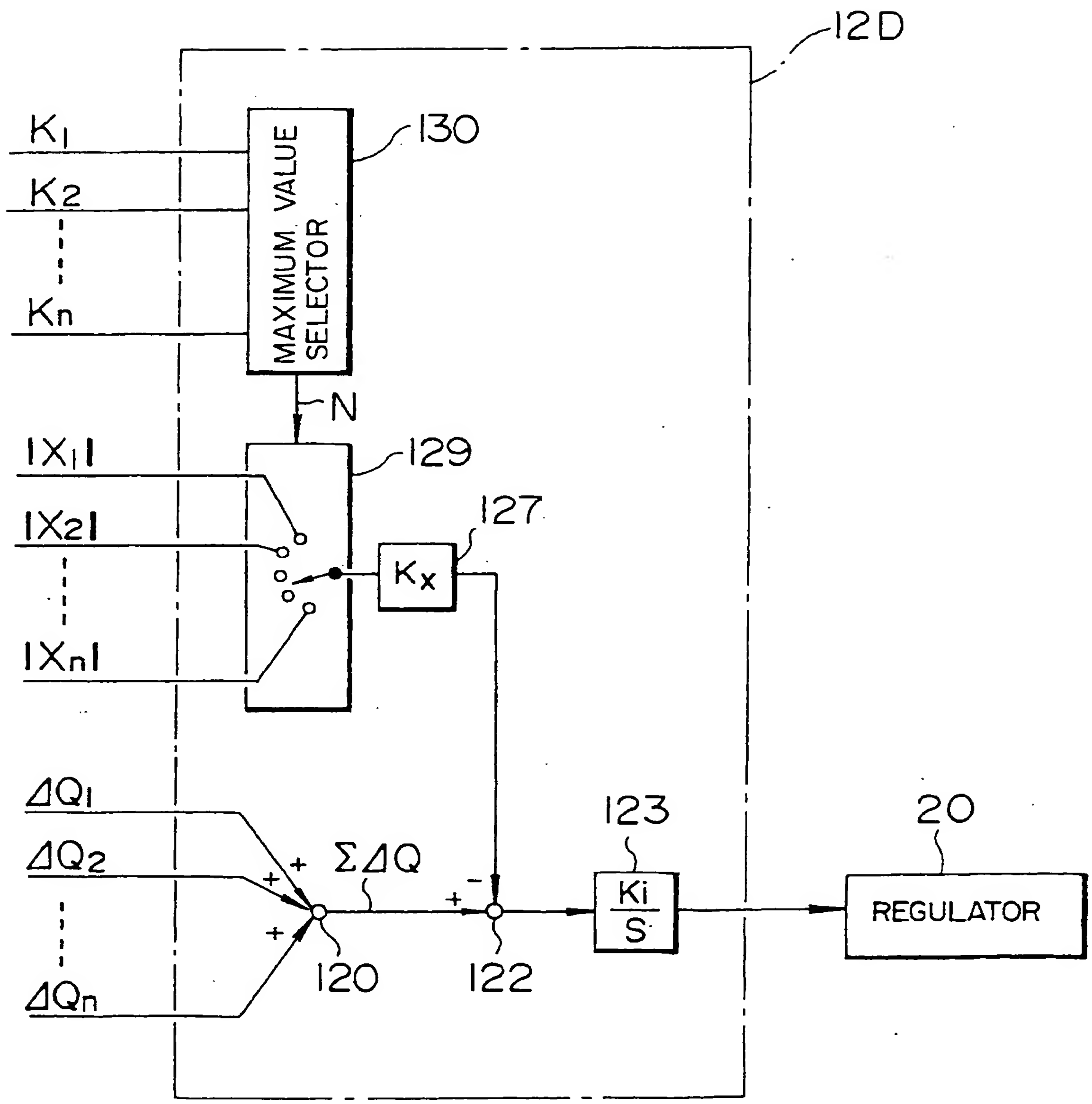

12D
K_1
K_2
K_n
MAXIMUM VALUE SELECTOR
130
N
$|X_1|$
$|X_2|$
$|X_n|$
129
127
K_x
ΔQ_1
ΔQ_2
ΔQ_n
$\Sigma\Delta Q$
+
+
+
+
−
120
122
123
$\frac{Ki}{S}$
20
REGULATOR

FIG.10

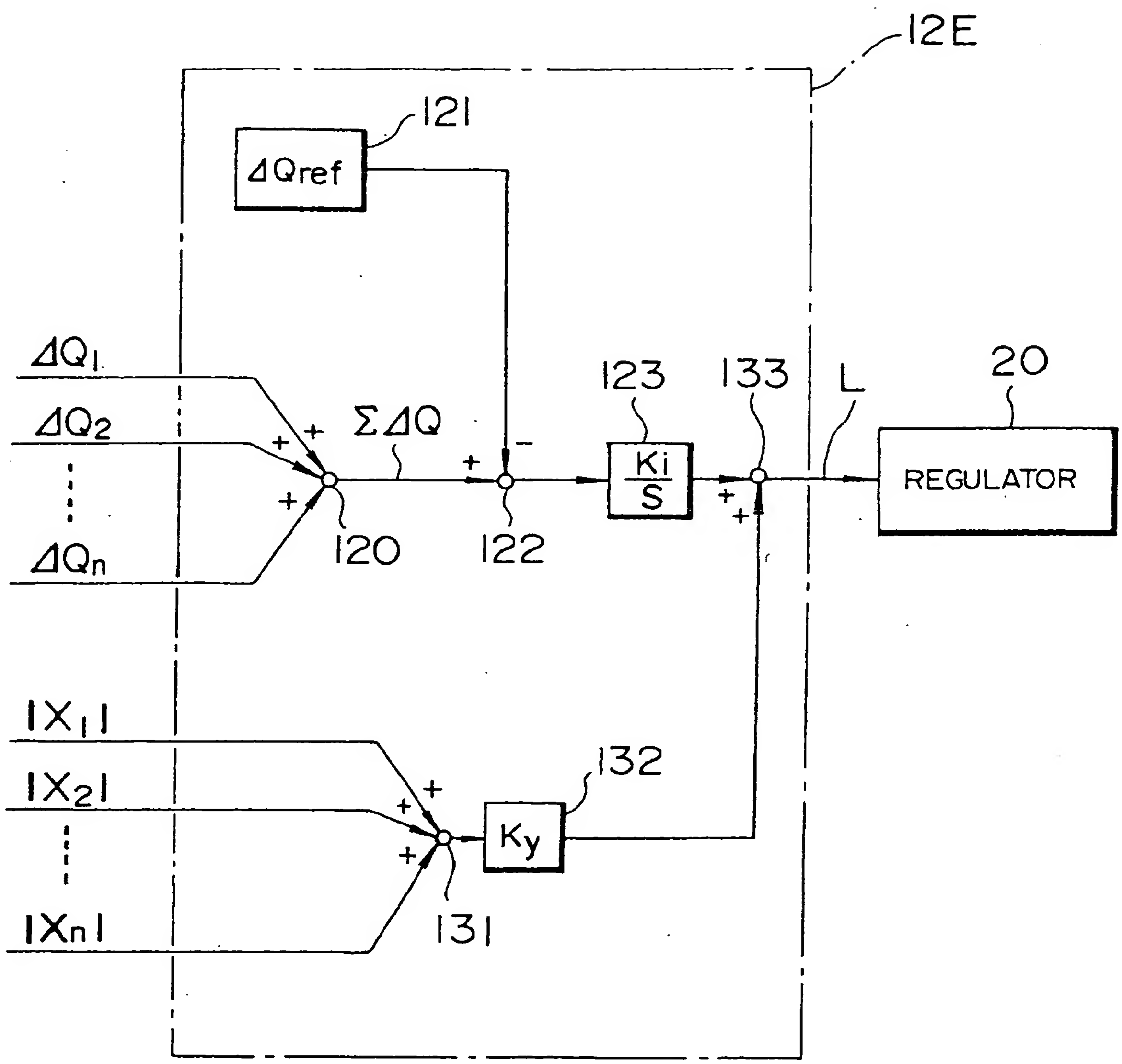

12E
121
ΔQref
ΔQ1
ΔQ2
ΔQn
ΣΔQ
+
+
+
+
−
120
122
123
Ki
S
133
L
20
REGULATOR
+
+
|X1|
|X2|
|Xn|
+
+
+
131
Ky
132

FIG. 11

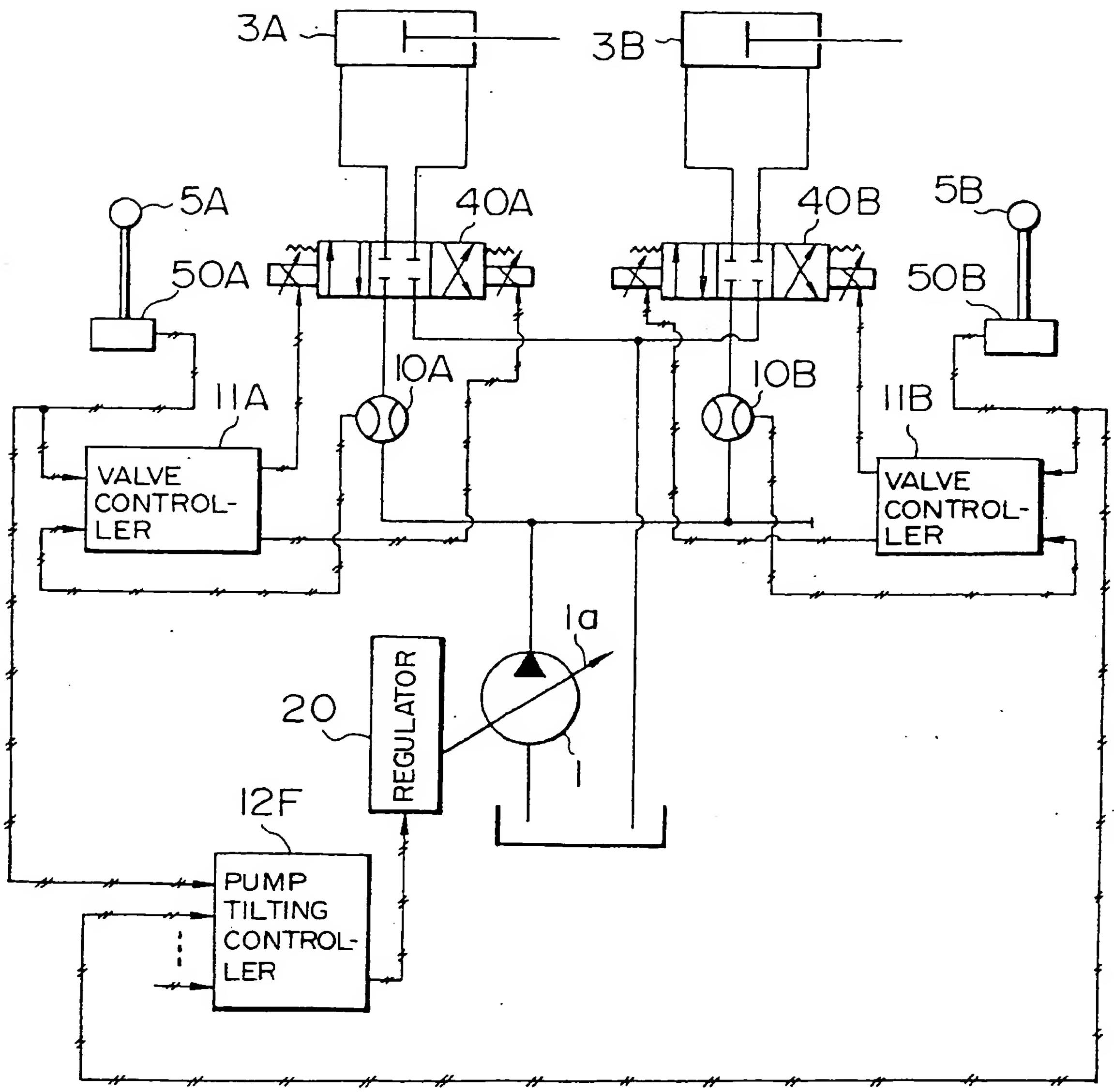
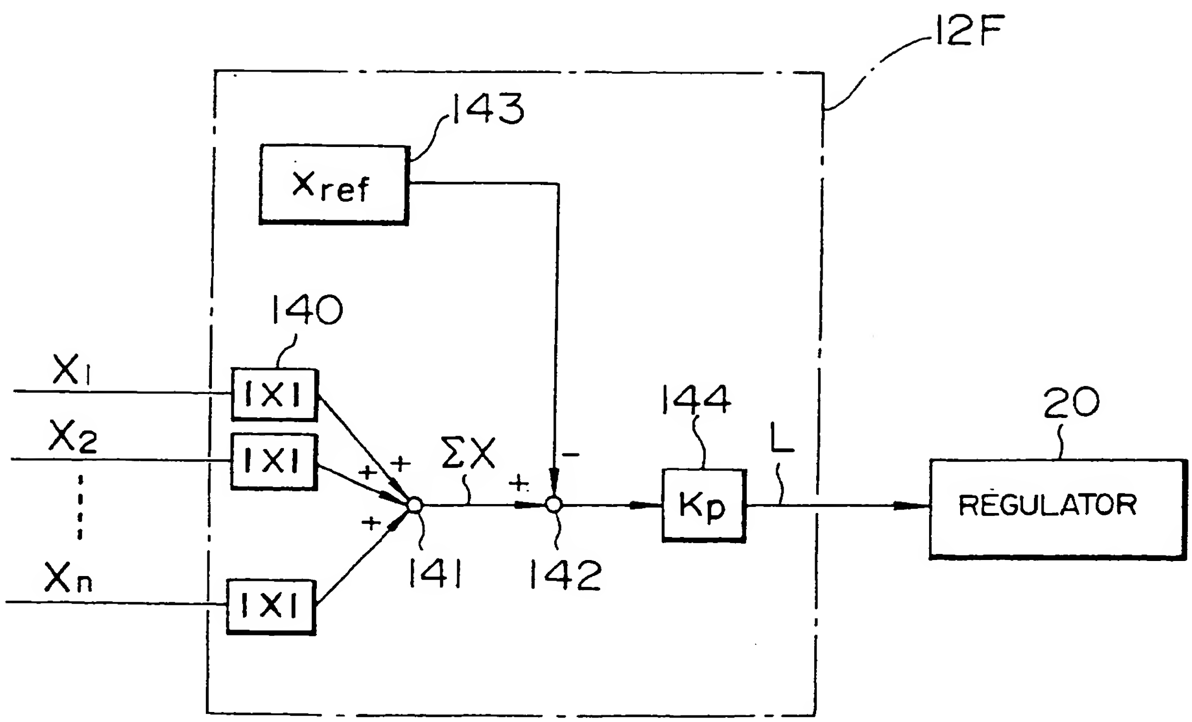

3A
3B
5A
40A
40B
5B
50A
50B
10A
10B
11A
11B
VALVE CONTROL-LER
VALVE CONTROL-LER
1a
20
REGULATOR
1
12F
PUMP TILTING CONTROL-LER

FIG. 12



INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP93/00197

A. CLASSIFICATION OF SUBJECT MATTER

Int. Cl⁵ F15B11/00, F15B11/05, E02F9/22

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

Int. Cl⁵ F15B11/00, F15B11/05, E02F9/22

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Jitsuyo Shinan Koho	1926 - 1992
Kokai Jitsuyo Shinan Koho	1971 - 1992

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP, A, 1-501241 (Caterpillar Inc.), April 27, 1989 (27. 04. 89), & US, A, 4712376	1-12
A	JP, A, 3-66901 (Komatsu Ltd.), March 22, 1991 (22. 03. 91)	1-12
A	JP, A, 63-120901 (Hitachi Construction Machinery Co., Ltd.), May 25, 1988 (25. 05. 88)	1-12

☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.

* Special categories of cited documents:

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"E" earlier document but published on or after the international filing date

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"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
March 4, 1993 (04. 03. 93)	March 23, 1993 (23. 03. 93)

Name and mailing address of the ISA/	Authorized officer
Japanese Patent Office	
Facsimile No.	Telephone No.